AD-A283 946

NPS-PH-94-009

NAVAL POSTGRADUATE SCHOOL
Monterey, California

IMPROVED EFFICIENCY AND POWER DENSITY FOR THERMOACOUSTIC COOLERS

by

Thomas J. Hofler

Annual Summary Report: Oct 1992 - Sep 1993

June 1994

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Prepared for: Office of Naval Research
Arlington, VA 22217-5660

NAVAL POSTGRADUATE SCHOOL
Monterey, California

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*

2. REPORT DATE
June 1994

3. REPORT TYPE AND DATES COVERED
Summary 01 Oct 92 - 30 Sep 93

4. TITLE AND SUBTITLE
Improved Efficieny and Power Density for Thermoacoustic Coolers

5. FUNDING NUMBERS
PE 601153N
G N0001493WR24060 &
N0001493WR23043
TA 3126976ess01 & ess02

6. AUTHOR(S)
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7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
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8. PERFORMING ORGANIZATION REPORT NUMBER

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
Office of Naval Research
ONR 331
800 North Quincy Street
Arlington, VA 22217-5660

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES
Annual Research Summary

12a. DISTRIBUTION/AVAILABILITY STATEMENT
Approved for public release;
Distribution is unlimited

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

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14. SUBJECT TERMS
Thermoacoustic, refrigeration, heat exchange, heat transport

15. NUMBER OF PAGES
22

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
unclassified	unclassified	unclassified	

Annual Summary Report for

Improved Efficiency and Power Density for Thermoacoustic Coolers

June 1994

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ABSTRACT

Research on improving the efficiency, cooling power, and cooling power density of thermoacoustic refrigerators is described. A heuristic analysis of short thermoacoustic heat exchangers in a high amplitude sound field is given. A heat exchanger experiment, utilizing a very high amplitude thermoacoustic prime-mover, shows some agreement with the heuristic analysis. This indicates that acoustic losses in the heat exchanger can be drastically reduced in high amplitude engines, while maintaining good thermal effectiveness. Other related, but more applied, research is briefly discussed. This includes the design and construction of a compact, portable, air-cooled, thermoacoustic refrigerator for the purpose of producing frost at a lecture demonstration. This design has roughly the same temperature span (40° C) as required by shipboard applications. Also, two new electrodynamic acoustic drivers have been designed and one design has been constructed. These designs offer high efficiency, good power density, and low cost and are probably scalable up to significantly higher acoustic power levels.

Improved Efficiency and Power Density for Thermoacoustic Coolers

TABLE OF CONTENTS

I Project Description

The goal of the proposed basic research is to consider and evaluate new thermoacoustic cooler designs which will lead to substantial improvements in both efficiency and cooling power density. The specific design aspects to be considered are the acoustic resonator, the stack, and the internal heat exchangers. Substantial improvements in both the efficiency and the cooling power density of thermoacoustic refrigerators is necessary in order for the technology to be competitive with existing vapor-compression systems.

II Approaches Taken

Reasonably high cooling power densities can be obtained if high acoustic amplitudes and high mean pressures can be used effectively. Acoustic amplitudes having a peak dynamic pressure that is at least 10% of the mean pressure are necessary ($p_o/p_m = 0.1$) and a figure of 20% or higher is desirable. While the acoustic loss for heat exchangers may be a modest fraction of the total, for low amplitude engines, it is much larger for high amplitude engines. For some high amplitude heat exchanger designs, the acoustic loss generated by both hot and cold heat exchangers could be the dominant source of loss in the entire system, or at least the largest single source of loss in the system. A simple, heuristic model describing the thermal effectiveness of thermoacoustic heat exchangers is described in section IV. These calculations indicated that very short heat exchangers having low acoustic loss can still be thermally effective, if they have the proper plate or fin separation.

Heat exchanger experiments have been conducted in order to test the heuristic model. These are described in section V. Although the goal of the project is to discover fundamental improvements in thermoacoustic refrigeration, the experiments were conducted with a thermoacoustic prime mover. The reason for this approach is that very large dynamic displacements of the working medium (gas) are essential, and achieving such large displacements with an electro-dynamically driven system is very difficult. The required electrodynamic drivers do not exist, and developing them is a substantial engineering effort.

Another approach we are taking on the problem, is to study the efficiency of the entire thermoacoustic resonator system including stack(s) and primary heat exchangers, using numerical models. An optimization of the parameters should be performed on a system wide basis, in order for the various compromises in component design to combine to form a meaningful system optimum. We have not made much progress on this as yet. It is our intent to build a smaller high-efficiency/high-power refrigerator, once a good numerical design has been finalized.

While the proposed research is to be basic in nature, its goals do overlap, in part, with our research on thermoacoustic cryocoolers. The cryocooler research is more applied in nature, but it has forced us to engineer a high-efficiency/high-power electrodynamic driver, in order to perform even the more basic aspects of our cryocooler research. In addition to enabling basic research, we believe that this driver design is an excellent one, in terms of low manufacturing cost and high efficiency, and could be used for some refrigerators of the type being considered here.

Finally, we have constructed a portable, air-cooled, low temperature span thermoacoustic refrigerator. Its primary purpose is to produce frost at lecture demonstrations. However, we may use it as a preliminary test bed for new refrigerator concepts.

III Summary of Completed Work

A simple heuristic theory for efficient thermoacoustic heat exchangers has been developed. This "short and narrow" heat exchanger model has been the subject of an oral presentation[1] and will be included in a forthcoming publication on understanding thermoacoustic heat transport in sub-boundary layer sized geometries. A first draft containing most of the heat exchanger theory material is given section V. While this theory is not precisely quantitative, it does provide rough quantitative estimates for efficient heat exchanger geometries. A simplified statement of the results is as follows.

Short finned heat exchangers with $\xi_{hx}/\Delta x_{hx}$ in the range of 3 to 8 can be thermally effective as a source or sink of thermoacoustic heat transport if y_o/δ_κ is in the range of 0.75 to 0.5. The peak displacement amplitude of a gas parcel in the heat exchanger is given by ξ_{hx} , Δx_{hx} is the length of the heat exchanger, y_o is the half separation between adjacent fins or plates , and δ_κ is the thermal penetration depth. Furthermore, the acoustic loss or dissipation of the heat exchanger, is relatively less for $y_o/\delta_\kappa \cong 0.5$ than for smaller or larger separations. This particular heat exchanger geometry should be very "efficient" in terms of having good thermal coupling to the stack and low acoustic dissipation.

Heat exchanger experiments have been conducted in order to test the heuristic model. Preliminary work on this experiment was presented at the Denver ASA meeting[2] and was completed as a master's thesis[3] by Lt. Nelson Castro, USN, in December 1993. In this experiment involving a room temperature to liquid nitrogen temperature prime mover, three different heat exchanger lengths were tested while holding all other experimental parameters constant. The results were evaluated primarily in terms of the measured amplitude p_o/p_m , and in terms of $\xi_{hx}/\Delta x_{hx}$. Briefly, the best values of p_o/p_m were about 27% for the longest exchangers where

$\xi_{hx}/\Delta x_{hx} \cong 2$. Also, the best values of $\xi_{hx}/\Delta x_{hx}$ were between 4 and 5 for the shortest heat exchangers where p_o/p_m ranged between 15% and 19%. This last result is particularly significant since it shows that the heat exchanger length can be very much shorter than a peak displacement amplitude of the gas, and still perform its function effectively. Other key workers on this project were Prof. Anthony Atchley and David Gardner.

Two new electrodynamic drivers of a slightly improved, Phase II, design have recently been built. These drivers feature improved robustness; modularity for ease of disassembly, inspection, and repair; and increase piston stroke. While the "motor" of the Phase II driver is not particularly powerful or efficient, we have measured electroacoustic efficiencies in excess of 45% and it should deliver acoustic power levels up to 25 Watts. Also, it has low-mass piston assembly resulting in a wide usable power bandwidth extending up to 700 Hz.

Also we have designed a more advanced, Phase III, high-efficiency electrodynamic driver and have tested some of the more critical components. The Phase III design calls for acoustic output powers of at least 60 Watts and an electroacoustic conversion efficiency in the range of 80% to 90%. Custom magnets and coils are currently being fabricated, but we do not expect to have complete units finished for some time because this advanced driver project is not our highest priority. We are currently working on a patent disclosure for these driver designs and so the details will not be presented here at this time. Contributors to the driver projects have been Jay Adeff, Lt. Kevin Mode, USN, and Lt. David Monahan, USN.

The compact demonstration refrigerator design was numerically optimized and drafted and mostly fabricated by Lt. Brent Brooks, USN. The refrigerator was assembled and run for the first time very recently. Although detailed measurements have not been made as yet, the performance is roughly as anticipated. It produces frost and the intended 40° C external uninsulated temperature span. Master's student Lt. Todd Berhow, USN, is currently measuring its performance, testing some minor modifications, and packaging it for easy transport to lecture and demonstration sites.

IV Thermoacoustic Heat Exchanger Concepts and Theory

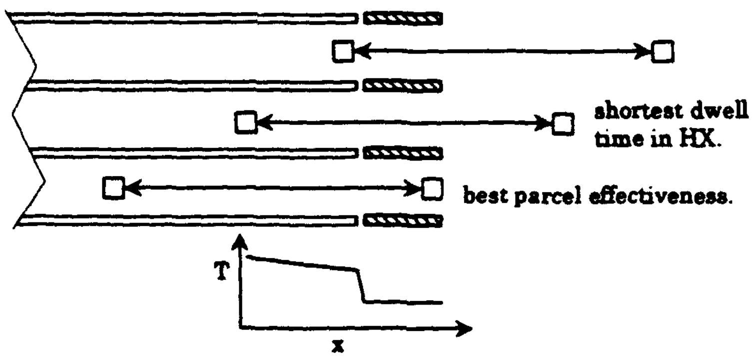
Swift[4] argues that any gas parcel that spends a short amount of time adjacent to the heat exchanger plate, relative to an acoustic period, does not make effective thermal contact with the heat exchanger plate. This is because Swift assumes that a thermal boundary layer of gas exists between the parcel and the plate, and that this boundary layer represents too large of a thermal impedance for a significant amount of heat to diffuse through the boundary layer in such a short time. One heat exchanger design that results from these boundary layer considerations is a heat exchanger that has the usual plate or fin separation of several penetration depths, and a length that is approximately equal to twice the peak displacement amplitude ξ_o, of a gas parcel. That is, $\Delta x_{hx} \cong 2\xi_o$, where Δx_{hx} is the length of the heat exchanger. This could be called the "maximum thermal effectiveness" design, since any gas parcel that makes any thermal contact with the stack will always be in thermal contact with the heat exchanger when it leaves the stack.

The problem with this design or geometry is that any engine incorporating a small temperature span stack, and large acoustic amplitudes for high power density, will have hot and cold heat exchangers whose total length is comparable to the length of the stack. Since the acoustic dissipation of the heat exchanger is significantly higher per unit length than it is for the stack, the total heat exchanger dissipation will likely be the dominant source of inefficiency for the entire system.

However, we propose a simple heuristic model[1] that suggests that a heat exchanger can be both thermally effective and short in the longitudinal direction $\Delta x_{hx} << 2\xi_o$, if the plate separation, in the transverse direction, is comparable to a thermal penetration depth $2y_o \approx \delta_\kappa$. By drastically reducing the heat exchanger length relative to the gas displacement amplitude, the surface area and acoustic dissipation should also be reduced.

The model begins with a number of simplifying assumptions. Gas viscosity and compressibility are ignored as are the details of the thermal contact between the stack and the gas parcels. The latter issue is complicated by the fact that the stack has a longitudinal temperature distribution T(x). The thickness of the plates are assumed to be negligibly small. Referring to Fig. 1, we will also assume that there is a discontinuity in the temperature T(x), between the end of the stack and the beginning of the heat exchanger plates. Furthermore, we will focus our attention on the second gas parcel from the top of Fig. 1, which is the parcel with the shortest dwell time t_D, in the heat exchanger. This is the shortest dwell time because the parcel has its peak velocity at the center of its excursion. Approximately $t_D \approx \Delta x_{hx}/\omega\xi_o$. This particular parcel is important because it has the poorest thermal contact with the heat exchanger. Finally, we assume that the heat exchanger length, in the *x* direction, is much greater than the fin separation. That is, $\Delta x_{hx} >> 2y_o$. This assures that the standard heat conduction solution for an infinite "slab" of material (see Fig. 3) is a reasonable solution for the gas in the exchanger.

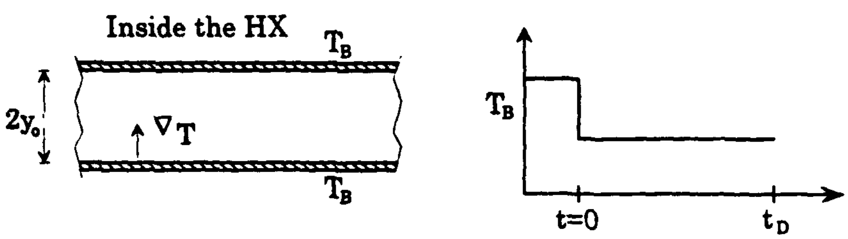
Figure 1 - Gas parcel movement and temperature distribution for short and narrow heat exchangers.



The question of thermal effectiveness can be stated thus: How long does the parcel have to remain in the heat exchanger to thermally equilibrate with the exchanger plates? It is important to note that perfect thermal equilibration is probably not a sensible goal, because this will require long heat exchangers, such as those in Swift's analysis. Long heat exchangers are undesirable because of the high acoustic dissipation incurred. A more sensible goal might be 90% thermal equilibration. That is, the final temperature of the gas parcel after passing through the exchanger is different from that of the exchanger by an amount that is only 10% of the temperature discontinuity in T(x).

We can recast the problem from one of a static temperature distribution in the plates and moving gas parcels, to one of a stationary slab of gas having a time dependent temperature at its boundaries. In Fig. 2, we assume that the temperature of the slab boundaries has a step discontinuity in temperature at time t=0 and we let the gas thermally equilibrate for a time equal to the dwell time t_D.

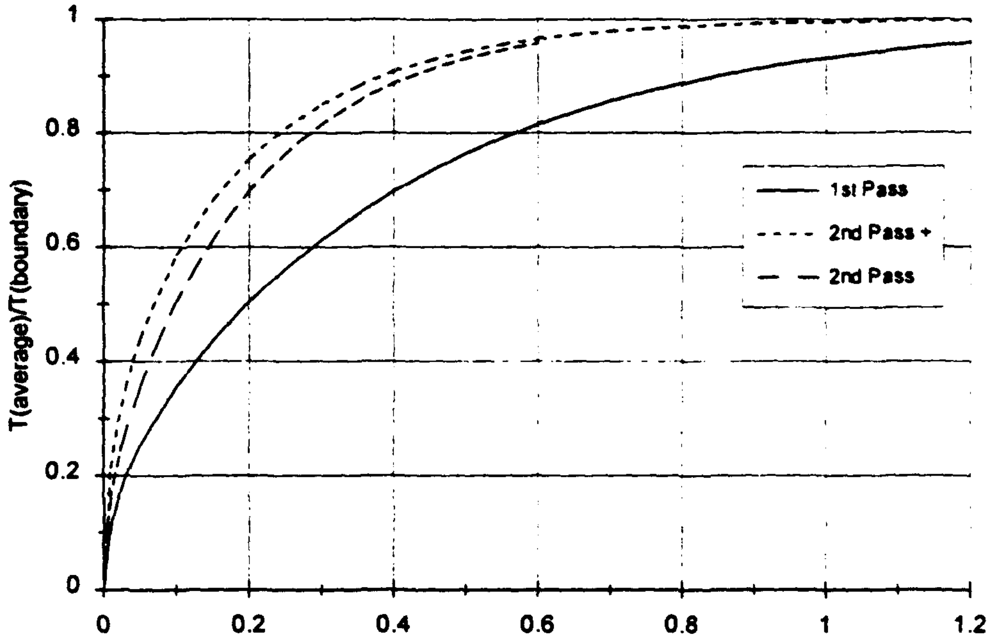
Figure 2 - Stationary thermal diffusion problem with a step function in the temperature at the boundaries.



The solution for the ratio of the gas temperature to the boundary temperature, averaged over the slab, can be found in a standard text[5]. This solution is indicated by the solid curve in Fig. 3, where a y-axis value of 1.0 represents perfect equilibration, and the x-axis values are normalized time $\tau = \chi t_D / y_o^2$, where χ is the thermal diffusivity. If we substitute our approximate value for the dwell time into the expression for τ, and we use the definition for the thermal penetration depth $\delta_\kappa = \sqrt{2\chi/\omega}$, we obtain

$$\tau \cong \left[2\left(\frac{\xi_{hx}}{\Delta x_{hx}}\right)\left(\frac{y_o}{\delta_\kappa}\right)^2\right]^{-1} .$$

Figure 3 - Average thermal equilibration of medium as a function of normalized time.



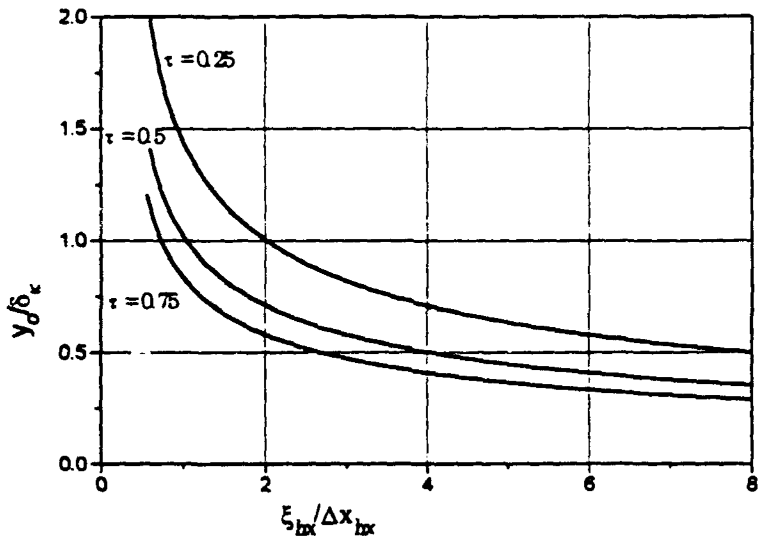
$$\tau = \frac{\chi t_D}{y_o^2} \cong \left[2\left(\frac{\xi_{hx}}{\Delta x_{hx}}\right)\left(\frac{y_o}{\delta_\kappa}\right)^2\right]^{-1}$$

The solid curve represents the amount of equilibration after one pass through the exchanger and the long dash curve represents the amount of equilibration after two passes (one outgoing and one incoming), in one acoustic cycle. This assumes that the temperature of a gas parcel is static when it is not in contact with either the stack or the heat exchanger. In fact, the temperature distribution of the gas parcels may be stratified in the transverse direction after one pass through the exchanger and they will be thermally equilibrating with their nearby neighbors. The short dash curve of Fig. 3 assumes perfect equilibration of the gas with itself, in between the first and second passes. The true solution will lie somewhere between the two dashed curves.

Note some values of equilibration versus τ. For $\tau = 0.25$, the equilibration is about 77%; for $\tau = 0.5$, the equilibration is about 93%; and for $\tau = 0.75$, the equilibration is about 98%. The curves shown in Fig. 3 represent the worst case gas parcels with respect to thermal equilibration and we may expect that the difference from perfect equilibration is as much as a factor of two better when averaged over all possible gas parcels. Thus values of τ between 0.25 and 0.5 may be a good choice for a heat exchanger, since this should guarantee a thermal equilibration in excess of 85%.

In Fig. 4, a range of parameters for y_o/δ_κ and $\xi_o/\Delta x_{hx}$ is bounded by the area between the $\tau = 0.25$ and $\tau = 0.5$ curves, for which the worst case thermal effectiveness is between 77% and 93%. Heat exchangers with $\xi_o/\Delta x_{hx}$ in the range of 4 to 8 can be thermally effective if y_o/δ_κ is in the range of 0.75 to 0.5.

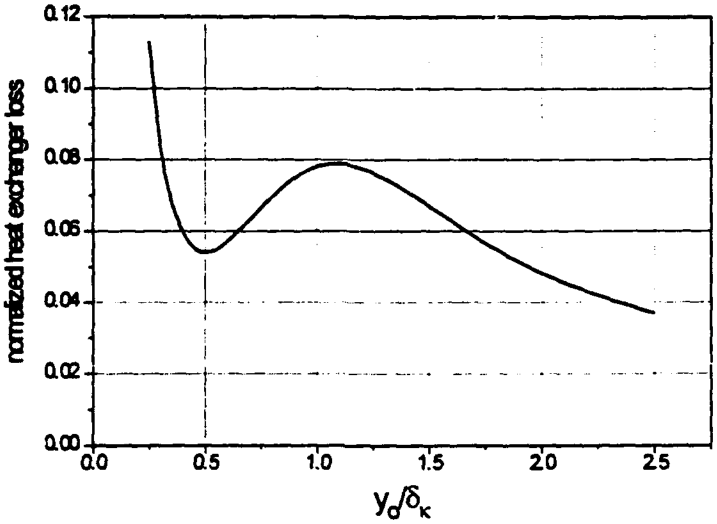
Figure 4 - Short heat exchanger parameter ranges for three different levels of thermal effectiveness.



We have not yet given any consideration to the actual acoustic dissipation of the heat exchanger. While the above model for thermal effectiveness is a large or finite amplitude model, we will only look at the usual infinitesimal amplitude theory for "thermo-viscous" losses. It must be noted that the physics of finite amplitude viscous dissipation in these heat exchangers is quite complex and these details are simply ignored here.

In Fig. 5, we calculate the normalized "thermo-viscous" losses for a specific hot heat exchanger in the "demonstration" refrigerator as a function of y_o/δ_κ. This particular refrigerator design is optimized for small temperature spans and this heat exchanger might be roughly representative of any small temperature span thermoacoustic refrigerator. The distance of the exchanger from the pressure antinode is $kx = 0.11$ radians, and the Prandtl number is 0.667. Also note that these losses should scale in direct proportion to the length of the exchanger as long as the acoustic pressure or velocity does not vary appreciably over its length.

Figure 5 - Acoustic dissipation in heat exchanger at kx = 0.11 radians, and Prandtl = 0.667.



Note the general hyperbolic trend of low loss at large values of y_0/δ_κ and high loss at small values of y_0/δ_κ, in Fig. 5. This is because the perimeter, or surface area per unit length, of the exchanger is inversely proportional to the plate separation. The features that are really interesting though, are the local maximum in exchanger losses at $y_0/\delta_\kappa = 1.1$ and the local minimum at $y_0/\delta_\kappa = 0.5$. The reason for the minimum at $y_0/\delta_\kappa = 0.5$ is because gas compressions are largely isothermal at this separation value, but peak velocities and viscous losses increase dramatically for even smaller separation values. If this dissipation result is combined with the above thermal effectiveness results, then it is apparent that some appropriate "figure of merit" function used in optimizing heat exchanger geometries would peak strongly at a value of about $y_0/\delta_\kappa = 0.5$.

While we do not expect these acoustic dissipation results to accurately represent the actual finite amplitude problem, we expect that the qualitative character is similar. Also, other positions of the exchanger (in the x direction) and other values of Prandtl number will change the character of Fig. 5 somewhat.

V Heat Exchanger Experiments

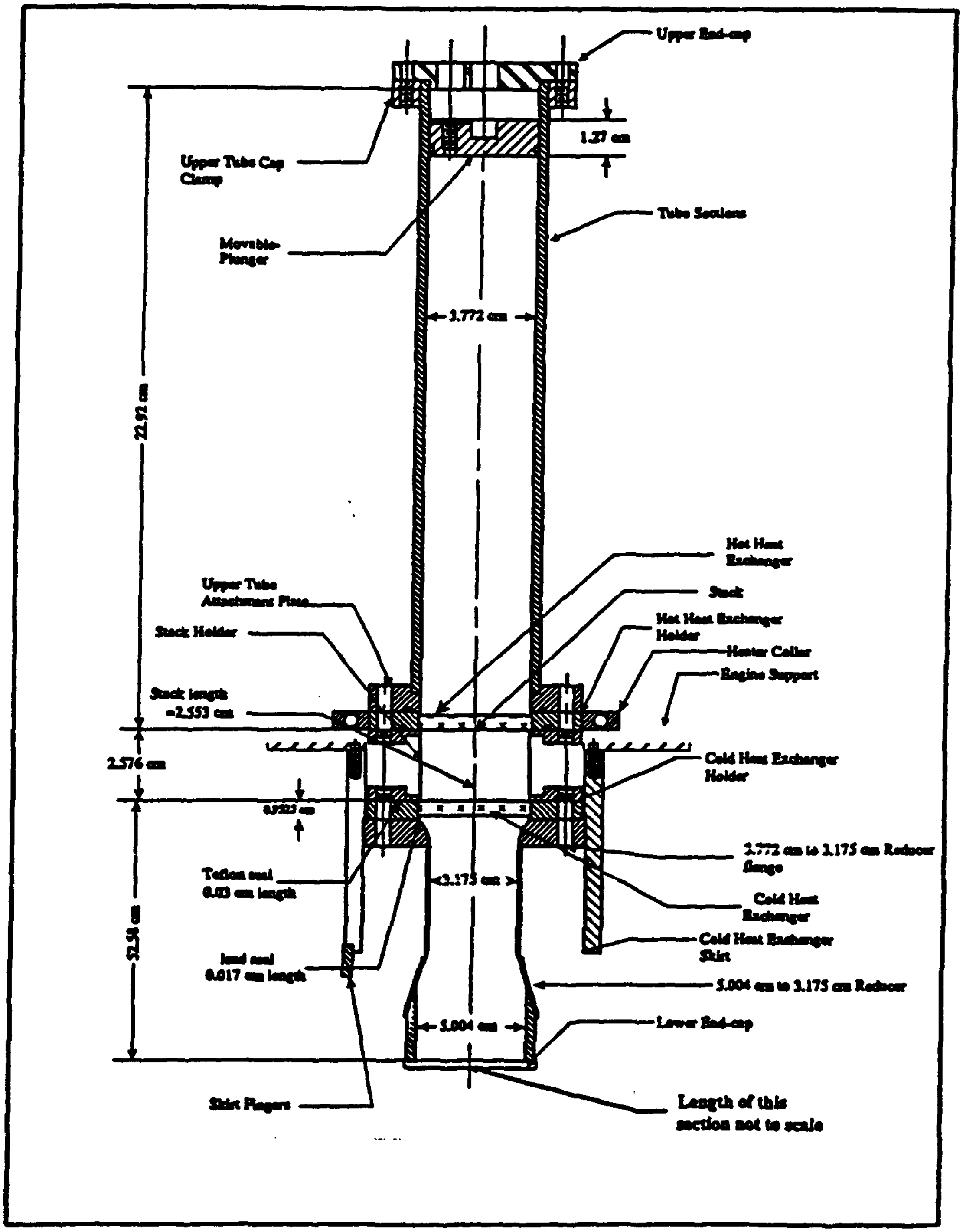
The objective for the heat exchanger experiment was to test and evaluate different thermoacoustic heat exchanger geometries in an acoustic environment having very large displacement amplitudes. A room temperature to liquid nitrogen temperature prime mover system was chosen instead of an electrically driven refrigerator because very large displacement amplitudes are difficult to achieve with electrodynamic drivers. High temperature heaters were avoided so that soft solders and plastics could be used in the construction; hence liquid nitrogen temperatures were used at the cold end.

An important requirement for the experiment was to suppress the heat flow density in the stack and heat exchangers so that uniform and constant temperatures in the heat exchangers could be maintained. Otherwise, the simple copper fin exchangers we used would not carry enough heat. A gas with a low sound speed and low mean pressure was used to reduce the heat flow at a given relative acoustic amplitude (eg. p_o/p_m). Unfortunately, at liquid nitrogen temperatures, neon is the heaviest monatomic gas that does not liquefy.

The method used in the experiment was fairly straightforward. We attempted to keep all aspects of the experiment constant except for three different heat exchanger pairs that were swapped in and out of the apparatus. Performance was evaluated primarily in terms of measured amplitudes, p_o/p_m. Given a fixed resonator geometry and temperature spans, heat exchangers that are thermally more effective will generate higher amplitudes. Similarly, everything else being equal, heat exchangers with lower acoustic losses will also generate higher amplitudes. An electric heater with a temperature controller was used to keep the hot heat exchanger near room temperature. While measurements of the heater power would have been interesting, we did not obtain reliable heater power measurements until near the end of this sequence of measurements. This data will not be presented here. Also, the stack and exchangers were extensively instrumented with thermocouples.

The apparatus is shown in Fig. 6, not to scale. The bottom portion of the resonator vessel is considerably longer than shown. The upper portion of the tube (3.77 cm inner diameter) features a sliding plunger with O-ring, so that the position of the stack in the standing wave can be adjusted. A single absolute pressure transducer is installed in the plunger which measures both the dynamic pressure via the AC component of the signal, and the mean pressure via the DC component of the signal. A heater collar is coupled to the hot heat exchanger flange. The stack is housed in a thin wall stainless steel tube. The cold heat exchanger flange is coupled to both the resonator tube and a large copper skirt with copper fingers, all of which are immersed in liquid nitrogen. The tube bore reduces to an inner diameter of 3.18 cm just below the cold exchanger and increases to an inner diameter of 5 cm closer to the bottom end of the tube. These diameter changes were required to obtain a relatively distortion free waveform. The total internal length of the gas column was approximately 65 cm.

Figure 6 - Section of the resonator.



The experiment was run with three different pairs of copper fin heat exchangers with three different lengths in the *x* or wave propagation direction. Exchangers in a pair were identical. All exchangers had a plate thickness of 0.254 mm and a plate separation of 0.508 mm. The three pairs had lengths of 0.257 cm, 0.569 cm, and 0.82 cm. The stack was a spiral roll of plastic film having layer separations of 0.85 mm. Two plunger positions of 8.13 cm and 10.7 cm relative to the hot heat exchanger were used, and mean neon pressures of approximately 8.7, 10.8, 27.5, 50 kPa were used.

The measured results are shown in Fig. 7 and Fig. 8, and some reduced results are shown in Fig. 9 and Fig. 10. Figure 7 shows the relative pressure amplitudes p_o/p_m, obtained as a function of mean pressure. Note that the longest exchangers produce the best amplitudes at the higher pressures, but that the shortest exchangers perform about as well as the longest at the lower 10.8 kPa pressure. Curiously, the medium length exchangers generally perform the poorest. Relative pressure amplitudes as high as p_o/p_m = 29% (not shown on the plot) were measured.

Figure 8 shows hot to cold temperature ratios measured at the center of the hot and cold heat exchangers where the temperature defect along the copper fins should be the largest. The ratios are plotted as a function of mean pressure where the prime mover is below onset at the lowest pressure, and above onset for pressures of 8.7 kPa and above. Note that our initial hope of constant and uniform temperatures is only roughly valid at the lower mean pressures and badly violated at the higher pressures. In spite of the low pressure neon, heater powers in the range of 100 to 200 Watts were measured at the higher mean pressures because of the extremely large amplitudes.

Two other features are worth noting in Fig. 8. The anomalously low relative amplitudes shown in Fig. 7 for the medium length exchanger (0.57 cm long) are explained in Fig. 8 where the temperature ratio is shown to be the lowest for this particular exchanger. In other words, the thermoacoustic "drive" was reduced. The reason for the low temperature ratio is hard to explain. Our best guess is that there were "cold" or imperfect soft solder joints coupling the finned exchanger elements to the flange(s) of the apparatus. While the solder joints appeared to be fine visually, something obviously added to the thermal impedance of the connections. The temperature ratio of the shortest exchanger was somewhat reduced compared to the longest exchanger, as is to be expected, since it simply has less copper in it. Note that the shortest exchangers are at a "unfair disadvantage" compared to the longest exchangers because of this reduced temperature ratio.

Figure 9 shows calculations of y_o/δ_κ for gas in the hot and cold exchangers as a function of mean pressure. The calculation included not only the mean pressure dependence of the thermal penetration depth, but also the unintended temperature variation at the higher pressures. Comparison of these values with the conclusion of the theory in section IV, would indicate that the relative plate separation values y_0/δ_κ, for the hot exchanger were good at the higher mean pressures, but too small at the lower mean pressures. The relative plate separation values for the cold exchanger were slightly too large at the lower mean pressures and very much too large at the higher pressures. In hind site, the cold exchangers should have had plate separations (in absolute units) that were a factor of 2 less than the hot exchangers. This would have placed the relative separations of both exchangers the "golden" range of about $y_0/\delta_\kappa = 0.5$ for mean pressures around 30 kPa. Also, t relative layer separations in the stack at the midpoint (half way between exchangers) ranged between $y_0/\delta_\kappa = 0.55$ at $p_m = 8.7$ kPa, to $y_0/\delta_\kappa = 1.25$ for $p_m = 50$ kPa.

Finally, Fig. 10 shows the measured amplitudes expressed in terms of calculated values of local peak displacement amplitude $\xi_0(x)$, relative to the heat exchanger and stack lengths, $\xi_0/\Delta x$. While the shortest heat exchangers did not produce the largest values of p_0/p_m, we can see that the shortest exchangers did produce peak displacement amplitudes that are locally as much as 5 times as large as the length of the cold heat exchanger. Another interesting feature of this plot are the values for the stack which in some cases exceeded $\xi_0/\Delta x = 0.5$. This indicates that the peak-to-peak displacement amplitude was greater than the length of the stack!

The extent to which these results clarify the problem of heat exchanger geometry is limited by a number of experimental problems which could be minimized or eliminated in the future. The largest problem is that a given pair of exchangers did not have approximately equal values of y_0/δ_κ, which could be fixed by fabricating more exchangers with different plate separations. Also, the variations in hot to cold temperature ratios should be minimized. But I think these measurements are good enough to show that heat exchanger lengths can be considerably shorter than a local peak displacement amplitude, and still function effectively. We hope to publish the heat exchanger theory and experiment material soon.

Figure 7 - Measurements of relative peak pressure amplitude vs. mean pressure.

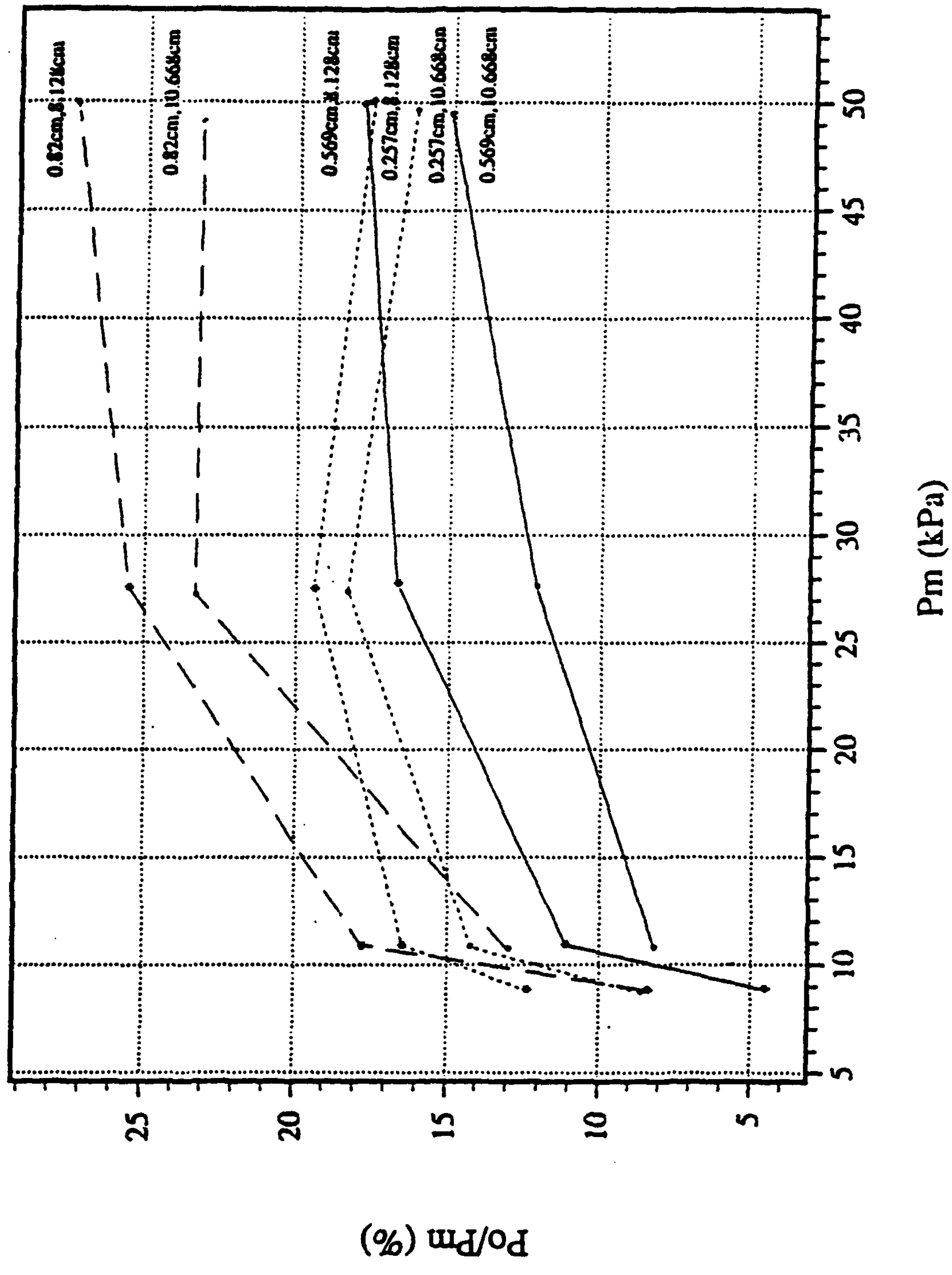


Figure 8 - Temperature ratio from hot exchanger to cold exchanger vs. mean pressure.

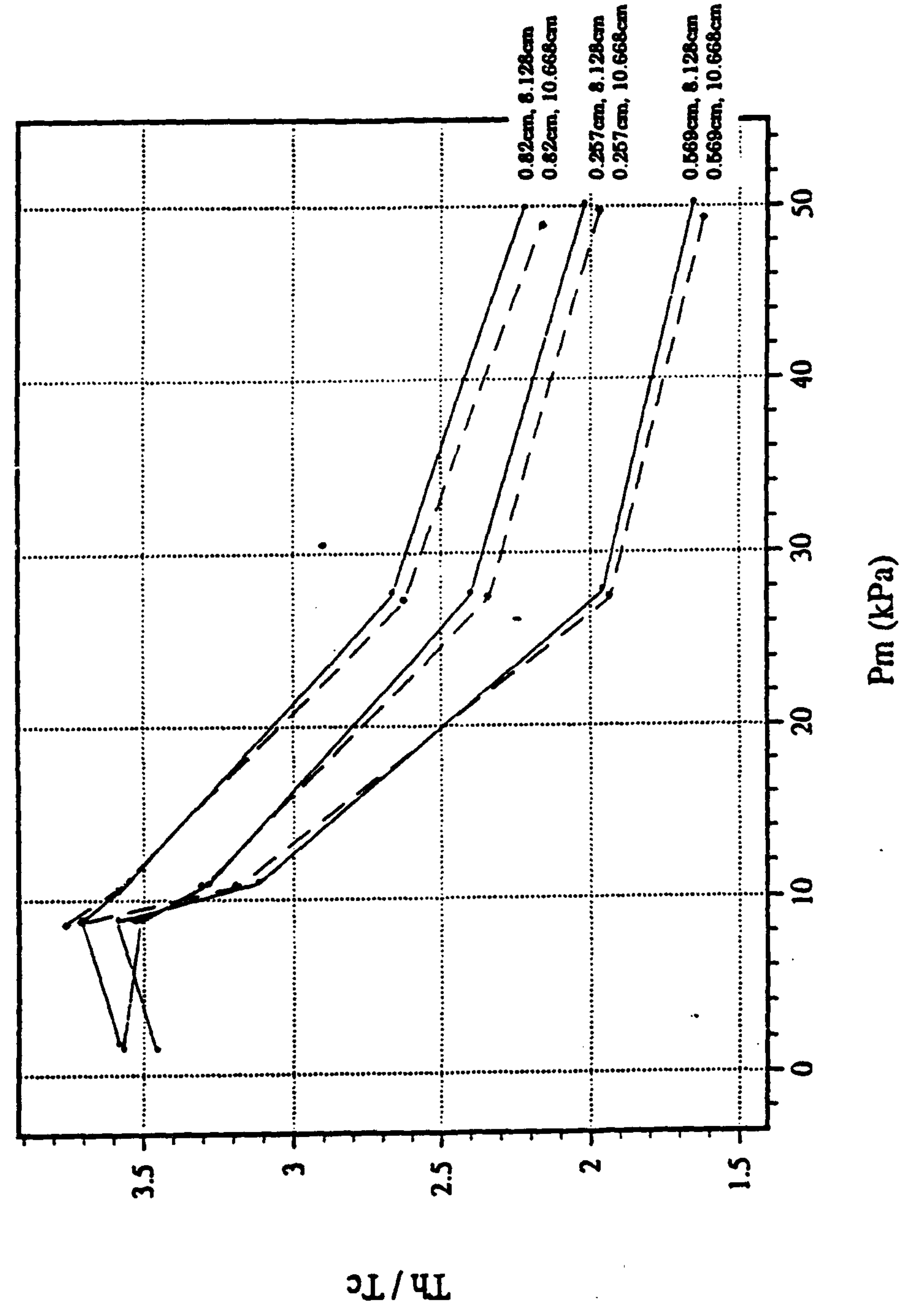


Figure 9 - Ratio of exchanger plate separation to local thermal penetration depth vs. mean pressure.

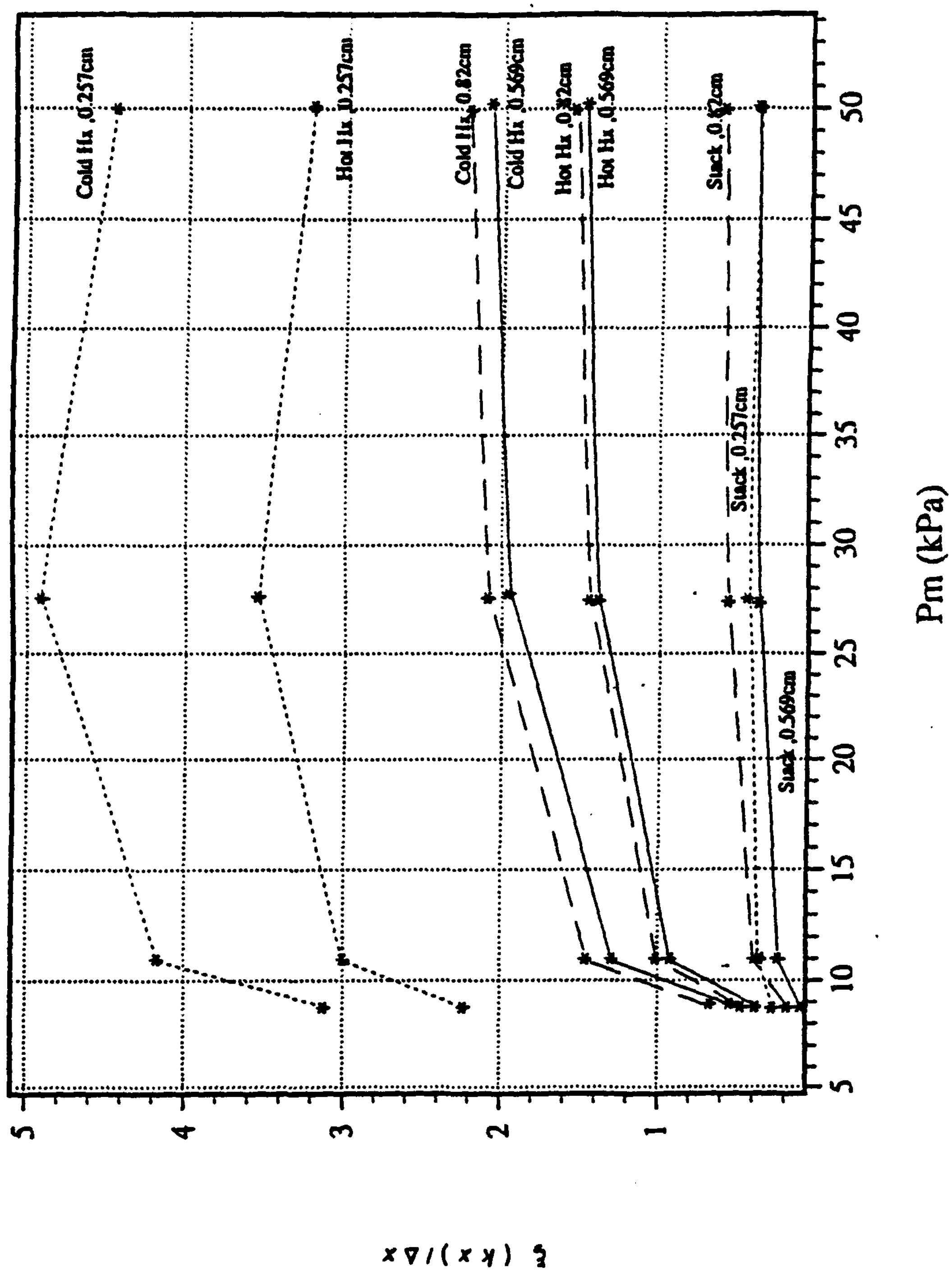
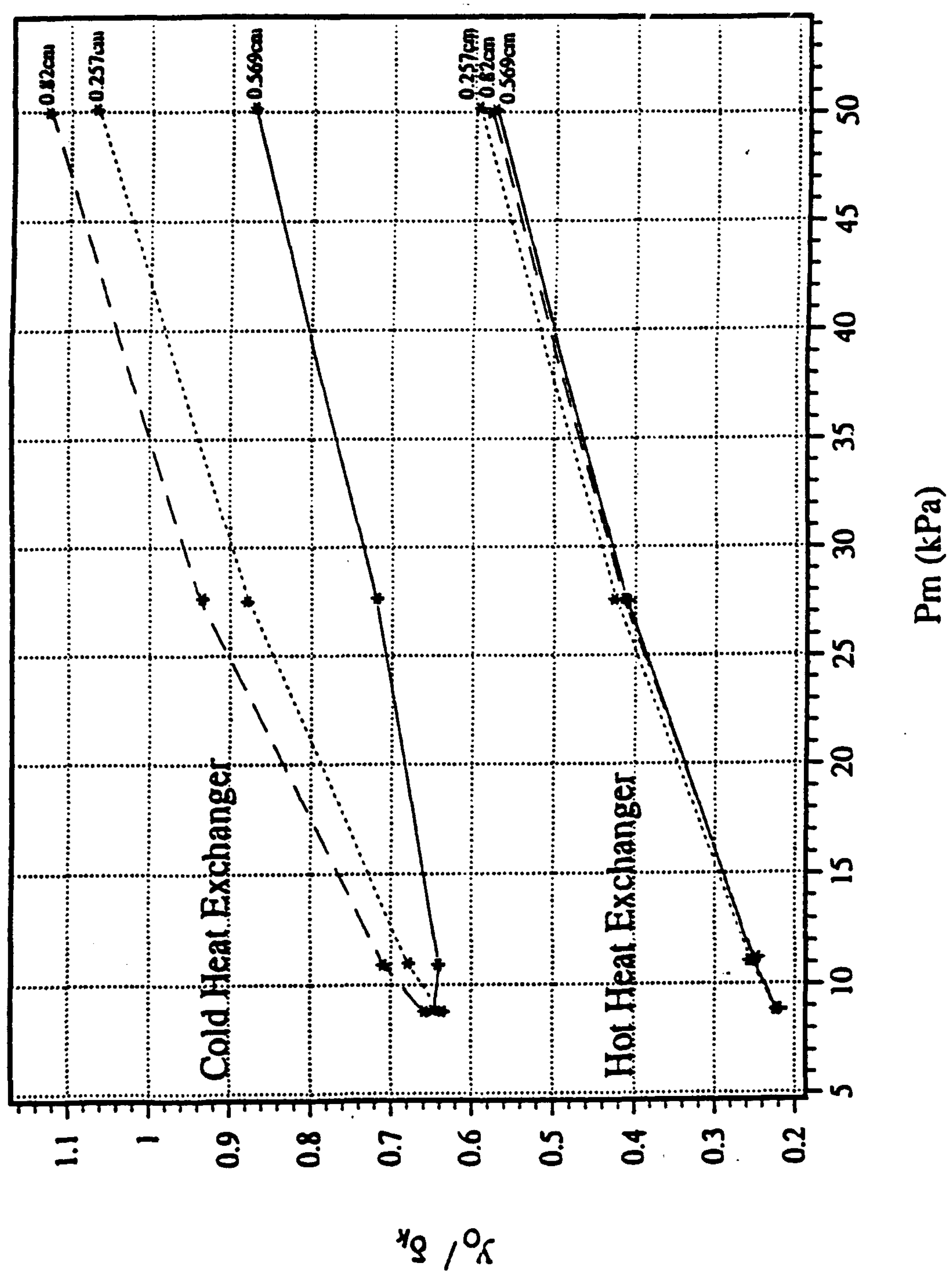


Figure 10 - Ratio of local peak displacement amplitude to stack or exchanger length vs. mean pressure.



VI Demonstration Refrigerator

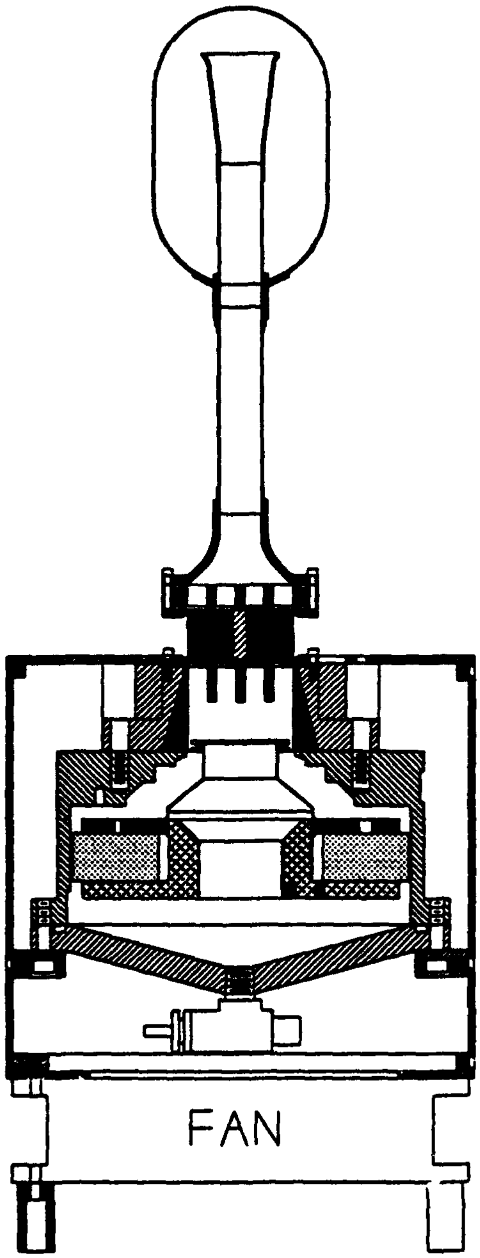
A little over a year ago we decided to construct a thermoacoustic refrigerator for demonstration purposes that was more advanced than previous demonstration hardware. The reasons were threefold: to have a more impressive refrigerator demonstration, to have a project that students could work on in the time frame of a Master's thesis, and to gain some experience with small temperature span refrigerators.

The basic design criteria were as follows. The refrigerator should be able to produce frost in an uninsulated configuration, and should be able to run for an indefinite period of time. Also, all of the equipment required to run the refrigerator should fit in a medium sized carrying case with a moderate weight. The specific constraints chosen from these general criteria were these: The hot end of the refrigerator and driver would be cooled with an integral fan. The internal (stack) temperature span would be 40° C, so that freezing temperatures could be obtained in spite of temperature drops at the hot and cold internal heat exchangers as well as for the final heat exchange to ambient air. The refrigerator would use our compact Phase II electrodynamic driver, which was the only choice readily available to us for a pressurized helium system. A mean helium pressure of 6.2 bar was chosen as offering good cooling power density and low strength requirements which allowed soft solder joints on the small diameter tube. A peak dynamic pressure of 5% of mean was chosen based on driver and overall power level considerations.

The entire system is modular, so that even the resonator and driver components can be broken down into sub components and reassembled rather quickly. Modified sub components could then be easily tested. Of course, certain modifications of a sub component might require the modification of other sub components.

A numerical model for the refrigerator was used to optimize several system parameters together with the above constraints. The resulting refrigerator is shown in Fig. 11. It is approximately the same overall design as the one in the Hofler patent[6] and the one used in the Space ThermoAcoustic Refrigerator, STAR[7]. Some details of the driver have been omitted from the figure.

Figure 11 - Section of a thermoacoustic demonstration refrigerator.



In order to shrink the size of the resonator we decided to run the helium filled system at 650 Hz, which is the highest operating frequency for the driver that gives reasonable driver efficiency. We also decided to use a smaller elongated float volume (shown at the top of Fig. 11), with a penetrating small diameter tube, for compactness. With the optimum resonator shape, the resonator was rather long. By reducing the diameter of the small diameter tube below the optimum, the resonator length was reduced by more than 2 inches, at a very small cost in efficiency. The length from driver to resonator float is about 14" and the overall length including fan is about 17."

The heat exchanger design is not quite as efficient as the one discussed in section IV because the fin separation is about a factor of two too large with a value of $y_o/\delta_\kappa \equiv 1.0$. The reason for this is that these finned exchanger are very difficult and time consuming to fabricate and we have not attempted extremely small spacings yet. Secondly, the exchangers that we used are already reasonably short and efficient because the acoustic amplitude is still rather low. The short exchangers each utilize three thick copper "cross-bars" or "thermal buss-bars" that are soldered to the fins at a 90° angle to the fins. The fins have most of the surface area and the cross-bars have most of the copper mass. The calculated temperature defect in the fins is about 0.5° C, which would have been about 10° C without the cross-bars. The cross-bars themselves have a temperature defect of about 2° C. The sum of all four temperature defects in the hot and cold exchangers is about 5° C. This does not include any defects between stack and exchanger or to ambient air temperature for the air cooling.

Although the refrigerator has recently been run and has produced frost and a measured <u>external</u> temperature span of about 40° C, no detailed measurements have been taken yet. However, the numerical model predicts a COP of about 1.2 which is about 20% of Carnot at a 40° C span. For a drive level of $p_o/p_m = 5\%$, the cooling power is about 12 Watts and the acoustic drive power is about 10 Watts.

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for
1 Oct 93 through 31 May 94

R&T Number: 3126976ess01 & ess02

Contract/Grant Number: N0001493WR24060 & N0001493WR23043

Contract/Grant Title: Improved Efficiency and Power Density for Thermoacoustic Coole

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a. Number of Papers Submitted to Referred Journal but not yet published: 0

b. Number of Papers Published in Referred Journals: 0
(list attached)

c. Number of Books or Chapters Submitted but not yet Published: 0

d. Number of Books or Chapters Published: 0
(list attached)

e. Number of Printed Technical Report & Non-Referred Papers: 0
(list attached)

f. Number of Patents Filed: 0

g. Number of Patents Granted: 0
(list attached)

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Grad Student Female	1
Grad Student Minority	
Grad Student Asian e/n	
Post-Doc Female	
Post-Doc Minority	
Post-Doc Asian e/n	

VIII Publications

Contributed Conference Presentations

Thomas J. Hofler, and J. A. Adeff "Performance of a thermoacoustic refrigerator with an improved stack geometry." J. Acoust. Soc. Am. Vol. 94, No. 3, Pt. 2, p. 1772, Sept. 1993, Denver Colorado.

Thomas J. Hofler, "Effective heat transfer between a thermoacoustic heat exchanger and stack." J. Acoust. Soc. Am. Vol. 94, No. 3, Pt. 2, p. 1772, Sept. 1993, Denver Colorado.

Nelson Castro, Thomas J. Hofler, and Anthony A. Atchley, "Experimental heat exchanger performance in a themoacoustic prime mover." J. Acoust. Soc. Am. Vol. 94, No. 3, Pt. 2, p. 1772, Sept. 1993, Denver Colorado.

Invited Presentations

Thomas J. Hofler, "Towards a Thermoacoustic Cryocooler," Bulletin of the APS Vol. 39, No. 2, p. 1085, April 1994, Crystal City, VA.

Thomas J. Hofler, "Thermoacoustic Refrigeration." Naval Postgraduate School, Physics Dept. colloquium, April 1994, Monterey, CA.

Thomas J. Hofler, "Thermoacoustic Refrigeration." Sacramento State Univ., Physics Dept. colloquium, Sept. 1993, Sacramento, California.

Supervised Theses

"Developement of a higher amplitude thermoacoustic driver having a relatively wide frequency bandwidth," Kevin S. Mode, Lieutenant Commander, USN, Master of Science in Engineering Acoustics, March 1994, thesis advisor.

"Design improvements for a high efficiency thermoacoustic driver," David A. Monahan, Lieutenant, USN, Master of Science in Applied Science, March 1994, thesis advisor.

"Construction of a thermoacoustic refrigerator demonstration apparatus," Brent R. Brooks, Lieutenant, USN, Master of Science in Applied Science, March 1994, thesis advisor.

"Experimental heat exchanger performance in a thermoacoustic prime mover," Nelson C. Castro, Lieutenant, USN, Master of Science in Physics., December 1993, thesis advisor.

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